N87-16748 D6-34
25 P.
49672

1986

NASA/ASEE Summer Faculty Fellowship Program

Marshall Space Flight Center

The University of Alabama

Advanced Thermal Energy Management:

A Thermal Test Bed and Heat Pipe Simulation

Prepared By:	Ronald G. Barile, Ph.D.
Academic Rank:	Professor and Program Chairman
University and Department:	Florida Institute of Technology Department of Chemical Engineering
NASA/MSFC: Lab	Structures and Propulsion
Division:	Engineering Analysis
Branch:	Thermal Engineering
MSFC Colleague:	James W. Owen
Date:	August 23, 1986
Contract No.:	NGT 01-002-099 The University of Alabama

D6-34

ABSTRACT

Work initiated in 1985 on a common-module thermal test simulation was continued, and a second project on heat pipe simulation was begun. The test bed is near completion and will be checked out in August 1986. Testing will involve various thermal conditions on the radiator, the cold plates, and the cabin heat exchangers. After the basic loop performance is established, other new technology items will be integrated with the loop to provide a flexible test environment for the new hardware. The first device will be a new type of heat pipe called a capillary pumped loop developed by CCS Associates sponsored by the SBIR program.

The test bed, constructed from surplus Skylab equipment, was modeled and solved for various thermal load and flow conditions. Low thermal load caused the radiator fluid, Coolanol 25, to thicken due to its temperature falling below -100 °F. The low temperature problem is best avoided by using a regenerator-heat-exchanger. Other possible solutions modeled include a radiator heater and shunting heat from the central thermal bus to the radiator. Also, module air temperature can become excessive with high avionics load.

A second project concerning advanced heat pipe concepts was initiated. A program was written (for an IBM-AT) which calculates fluid physical properties, liquid and vapor pressures in the evaporator and condenser, fluid flow rates, and thermal flux. The program is directed to evaluating newer heat pipe wicks and geometries, especially with external arteries. A case study is presented for water in an artery surrounded by six vapor channels. Effects of temperature, groove and slot dimensions, and wick properties are reported.

ACKNOWLEDGEMENTS

This work would not have been possible without the many warm and intelligent people in Structures and Propulsion Laboratory: Jim Owen, Bill Till, Joe Sims, Dick Wegrich, Farouk Huneidi, Tim Page, Ron Davenport, Jeff Bland, Al Shariett, Bill Patterson, Mickey White, Dave Clark, Dave Patterson; the managers: Alex McCool, Paul Frederick, and Bob Richmond; and, the essential secretaries: Betty Golden, Jeanette Hall, and special thanks to Sue Madole who typed my manuscripts both summers.

Appreciation is also extended to the program directors who faithfully attended the many seminars, meetings, and tours, and made the summer experience highly successful: Mike Freeman and Dinah Conrad, Ernestine Cothran, and Dr. Fred Speer.

INTRODUCTION

This work is in collaboration with Jim Owen and Dick Wegrich of the Thermal Engineering Branch of Structures and Propulsion Laboratory. The group is responsible for solving a wide variety of thermal management problems in space missions including radiation to and from satellites, most recently the Hubble space telescope, space shuttle, and space station, including habitat and lab modules, and experiment environmental control. The topics covered here include primarily space station thermal management technology.

Pressurized compartments in the space station, where astronauts live and work, will be subjected to various heat inputs (thermal loads). Internally, heat is dissipated to the cabin atmosphere from all electrical and mechanical equipment, computers, lights, experiments, refrigeration, and even human metabolism. Externally, the sun and earth will radiate energy to the station, while the outer walls will radiate to deep space. The external heat sources and sinks will vary seasonally, as well as differences between individual compartments due to shading and view factor variability. Life support, experiment, and other environmental temperature requirements are such that an active thermal management system will be designed for each module. The basic units in a thermal system are cold plates and air heat exchangers for gathering waste heat, pumped coolant and heat pipes for transporting heat, and radiators for rejecting heat. There is even consideration of storing heat in phase change materials such as waxes to reduce peak heating of the radiator and thereby reduce the size and weight of the radiator. Heat picked up from the interior will be rejected by body-mounted radiators or transferred to a central thermal bus. The central bus is an external fluid loop which will transport heat to the central radiators where heat is radiated to space.

During the summer of 1985, the author took preliminary plans from Jim Owen for a thermal test bed and did calculations, specified the loop, identified hardware, and made preparations for its construction by Lockheed Missile and Space Company, Huntsville. The loop was to represent the common module of space station in terms of heat loads and thermal management. The objective was to provide an early in-house test capability for thermal systems which supports the MSFC work package.

The test bed is near completion and will be checked out in August 1986. Testing will involve various thermal conditions on the radiator, the cold plates, and the cabin heat exchangers. After the basic loop performance is established, other new technology items can be integrated with the loop to provide a flexible test environment for the new hardware. The first device will be a new type of heat pipe called a capillary pumped loop developed by CCS Associates sponsored by the SBIR Program. Other hardware on the near horizon includes new radiators, thermal storage fluid, refrigerator, thermal bus, long-life fluid systems, rotating heat pipe joint, electrochromic panels, and metal hydride applications.

Thermal loop configuration, hardware specifications, and computer simulations were covered in the 1985 Summer Faculty Program report (4). For practical necessity, certain hardware changes have been made since that document was prepared: The thermal radiator was changed to an alternative with 24 sq ft surface; one of two cabin-air heat exchangers was eliminated; a thermal storage device was added (1650 Btu); all thermal loads were modified (solar input, cabin air, cold plates); and the number of heat exchanger sub-units in the radiator and facility heat exchangers was changed. The computer simulations presented here incorporate these latest changes and, to that extent, are different from those in the 1985 report.

A second major effort was begun this year to develop a heat pipe model for use on an IBM microcomputer. The model was to be flexible such that new geometries and new concepts could be screened by simulation. Then, promising designs could be specified for fabrication and experimentation. The motivation for improving heat pipes stems from space station requirements for thermal management which encompass tens-of-kilowatt heat loads to be moved over tens-of-meters distance. A serious limitation of current heat pipe technology is the small hydraulic pumping head which is developed in the wick at the vapor-liquid meniscus. Advanced heat pipes can pump fluids up to 0.1 psi total pressure drop, but the anticipated need for space station is on the order of 1 to 10 psi. Hence, this key problem of hydraulic pumping limit is currently under investigation, and the model and case studies presented here are an initial step. The alternative under consideration now is to actively pump the liquid to the heat loads, and provide liquid and vapor management hardware. Control of this two-phase thermal bus system is anticipated to be very difficult. The heat pipe offers the attractive alternatives of being completely passive, self-controlled, and self-contained.

OBJECTIVES

The ultimate objectives of this project are to develop a flexible thermal management test bed capable of evaluating advanced thermal systems for the space station common module, and to search for new concepts in heat pipe technology. The immediate objectives are:

1. Model and test the thermal loop under various load conditions.

2. Write a heat pipe program for a microcomputer and run test cases of new designs aimed at higher pumping capacity.

THERMAL TEST BED

A. CONSTRUCTION AND TEST PLAN

Much of the equipment for the test bed was already in storage at MSFC. Leftover from ATM and skylab are pumps, valves, sensors, filters, and heat exchangers (1). Design specifications of these elements are available (2). A double-sided radiator with 24 sq ft effective area and several cold plates are also available. The general configuration of these elements is shown in figure 1. Infrared lamps are to be situated around the radiators to simulate variable orbital conditions. The lamps are to provide 0-17.3 w/sq ft controllable, evenly distributed heating. This is equivalent to 0-1400 Btu/hr insolation for the specified radiators. Inside the radiators, a Coolanol 25 fluid will be used (Monsanto). The other loops will contain water. Interface heat transfer from the internal water to the thermal bus will be simulated by a facility water loop. General specifications and parameters for the test bed elements are shown in table I.

Lockheed Missile and Space Company, Huntsville, AL, has assembled and mounted the Coolanol and water loops on aluminum tables. Preliminary checkout will be done at Lockheed. Then the test bed will be shipped to building 4619 to prepare for tests in the Sunspot facility when it is available.

A description of the test facility requirements and of the tests is found in reference (1). The radiator will be situated inside the vacuum chamber. Cold plate loads and lamp power will be varied to simulate a range of internal and orbital conditions. As new technology concepts and hardware become available, they will be incorporated into the tests.

B. TEST BED FLOW SHEET

Nominal conditions, fluid flows, heat loads, and heat flows are shown in figure 2. Coolanol 25 is recommended for the radiator fluid. It has kinematic viscosity of 12 Centistokes and density of 0.91 g/cc at 0 °F. Half-inch ID tubing is adequate for this service. However, at -50 °F, it will thicken to 200 Cs and 0.95 g/cc which may make it unpumpable. The original fluid on skylab, Coolanol 15, is

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TABLE I

Equipment Specifications

Radiators: Double sided, total area = 24 sq ft. Approximate heat rejection up to 5200 Btu/hr. Coolanol 25 fluid.

Heat Exchanger 1: HX1 consists of four units in series, originally designed for ground service to airlock, ATM, and suit cooling. Original rating was 17,700 Btu/hr each, at 183 lb/hr each (UA = 258 Btu/hr - sq ft - F each). Present application: 1,700 Btu/hr at 183 lb/hr for each of four units. Part number 52-83700-1202, ref. (2), pg. 25.0.

Heat Exchanger 2: HX2 consists of a cabin air heat exchanger with water coolant. Original rating was 680 Btu/hr at 88 CFM, 5 psia oxygen (UA - 48.1 Btu/hr - sq ft - F). Present application: 1000 Btu/hr. Part no.: 52-83700-1227, ref. (2), pg. 32.0.

Heat Exchanger 3: HX3 consists of 2 units connected in series. The units are the same as HX1 units above. The present application is 7400 Btu/hr transfer from loop water (520 lb/hr) to facility water (550 lb/hr, 65 F). Facility water represents the central bus.

Cold Plates: Cold plates 1 through 5 are nominally 0.6 kw, or a total of 3 kw.

Pump Systems: The radiator pump system provides pump, accumulator, low fluid and power indicators, and a fill port. The pump is rated for "coolant" at 183 lb/hr with 175 psi pressure rise. part nos. 52-83700-831, 833, and 869, ref. (2), pp. 19.0-21-3. Water pump: 2 gpm; maximum pressure: 80 psi.

Radiator Mixer Valve: Details of the valve selection were included in a separate communication (5). Two options were considered: (a) Proportional-integral control using a proportional valve driven by a dc gear wound motor, and (b) stepper control using 5 parallel tubes fit with orifices of different openings and 4 on-off solenoid valves to select flow paths. Temporarily, a manual valve is installed.

less viscous but it is no longer made. Water flow at 520 lb/hr and 70 °F is handled by 1/4 or 3/8 inch ID tubing (6.8 and 3.0 ft/s, respectively.)

The lamps surrounding the radiator are designed for 0-17.3 w/sq ft which is 0-1400 Btu/hr total incident radiation on the upper surface. The lamps have a narrow profile and will not greatly affect outward radiation. An estimate of the total radiator rejection power is 1.5 kw or 5,200 Btu/hr. Part of the Coolanol will bypass the radiator, a fraction f, and mix with the cooled fluid as shown in figure 1. The mixed outlet design temperature is 40 F. The rejection heat load on the radiator is that absorbed from the lamps and the heat picked up in heat exchanger 1 (HX1). The cabin air cooling load is roughly 500 to 1000 Btu/hr. This plus cold plate (CP-1) load totals 3050 Btu/hr, the nominal load for HX1. Cold plates 1 through 5 have nominal 0.6 kw heaters; i.e., the total load for cold plates 1-5 is 3.0 kw or 10,245 Btu/hr. The heat taken up in cold plates 2-5 is primarily rejected in HX3 to facility water. This heat exchanger represents the space station central bus interface.

An extra heat exchanger, HX on figure 1, is planned for the situation where heat loads are greater than the radiator rejection capability. Up to 3500 Btu/hr may be rejected in HX, in which coolant service at 20 °F is required. This heat exchanger can also function as a ground service exchanger.

C. SIMULATION OF TEST BED PERFORMANCE

Equations describing heat transfer in the bed are given in table II. The 20 equations are listed in order of solution by TK!Solver:

Heat exchanger 2, Eqns. HX2 1-4

Radiator and control valve, Eqns. RAD 1-5

Heat exchanger 1, Eqns. HX1 1-4

Heat exchanger 3, Eqns. HX3 1-3

Cold plate 1, Eqn. CP1 1

Tee, Eqn. TEE 1

Cold plates 2-5, Eqns. CP2-5 1-2

<u>TABLE II</u>

TK!SOLVER MODEL EQUATIONS

```
        S Rule
        - ----
HX2   1 * q2=ww1*(tw2-tw1)
      2 * q2=fa*1*pa*0.6489*60*(ta1-ta2)/(ta1+460)
      3 * q2=48.1*dt2
      4 * dt2=((ta1-tw2)-(ta2-tw1))/ln((ta1-tw2)/(ta2-tw1))
RAD   1 * qrn=wc*cpc*(tc2-tc3)
      2 * qrr=qrn+qra
      3 * tr=(qrr/3.415/8.977E-10/ar)^0.25-460
      4 * tc1=2*tr-tc2
      5 * f=(tc3-tc1)/(tc2-tc1)
HX1   1 * q1=wc*cpc*(tc2-tc3)
      2 * dt1=q1/1032
      3 * tw4=q1/ww1+tw1
      4 * dt1=((tw4-tc2)-(tw1-tc3))/ln((tw4-tc2)/(tw1-tc3))
HX3   1 * q3=ww5*cpw*(tw6-tw4)
      2 * tf2=q3/wf+tf1
      3 * q3=516*((tw6-tf2)-(tw4-tf1))/ln((tw6-tf2)/(tw4-tf1))
CP1   1 * tw3=qcp1/ww1+tw2
TEE   1 * tw5=(ww1*tw3+ww2*tw4)/ww5
CP2-5 1 * qcp=qcp2+qcp3+qcp4+qcp5
      2 * qcp=ww5*cpw*(tw6-tw5)
```

Details of the thermal model and solution are given in reference (4). Changes in the completed hardware are reflected in the present calculations using the model in tables II and III. The new radiator model, equation RAD-3, was developed from a single known performance point of the radiator: 1700 watts rejected at 70 °F average temperature. The heat rejected, qrr, is simply a function of tr, the average radiator fluid temperature. Emissivity and efficiency are included in the constants. This model is not directly sensitive to changes in coolant flow rate. But, it will predict very cold outlet temperature when the coolant bypass rate is around 90%, i.e., when loads are small.

Design equations (Eqns. HX2-3, HX1-2, and HX3-3) contain an overall heat transfer parameter, UA, Btu/hr-F. Values for UA's in table II are:

HX2 UA = 48.1 Btu/hr - F

HX1 UA = 4 x 258 Btu/hr - F

HX3 UA = 2 x 258 Btu/hr - F

The cabin air load was assumed fixed at 500 or 1000 Btu/hr. The model will solve for cabin air temperature, ta1, and air outlet temperature, ta2. In this way, the value ta1 indicates whether or not the ECLS standard is being met. Facility water input to HX3 is assumed constant at 550 lb/hr, 65 °F. This will reasonably approximate the test situation but will not simulate a two-phase central bus. The model will require slight modification of the HX3 equations in the latter case.

During high-load situations, the radiator will not dissipate enough heat to retain tc3 at 40 °F. A signal for this condition is a value of f outside its range of 0 to 1. When this happens, the model and the variables must be changed slightly. The model equation RAD-5 is replaced by:

tc3 = tc1

The variable change is:

Input 0.0 for f

As a result of these changes, all the Coolanol will flow through the radiator and the outlet temperature (tc1 = tc3) will be calculated. This result will be above the set point of 40 °F. A typical output list from TK!Solver is given in table III for the case of 50% load and a cabin load of 1000 Btu/hr.

Several case studies were run to encompass a range of conditions to be tested. Figure 3 shows the results of loop simulation with TK!Solver for radiation and cold plate loads of 0, 25, 50, 75, and 100%. Temperatures at various positions are listed, as well as the Q1, Q2, and Q3 (heat transferred in HX1, HX2, and HX3) ranges. A load of 100% was calculated as overload with _f_ specified as 0.0. TC1 and TC3 show radiator outlet and mixer valve outlets. These two converge between 75 and 100% load. Beyond this point, they physically must converge because the radiator bypass is closed. Also, plotted are TW1, the lowest water temperature in the loop (inlet to cabin heat exchanger), TW6, the highest water loop temperature (outlet from cold plates 2-5), and TA1, the cabin air temperature inlet to HX2. Water and air temperatures are well behaved and within nominal ranges.

A problem at low load is indicated by the very low temperatures of TC1. The Coolanol fluid will become very viscous below -50 °F and probably become a gel-like fluid. This problem might be avoided by operating the mixer valve to permit no more than 75% bypass, i.e., $f \leq 0.75$. A "lowload" situation is achieved by simply setting $f=0.75$. This results in increased TC1 but water may freeze in parts of the loop.

Several schemes were simulated to improve the zero-load case inorder to bring TC1 above -50 °F. A 1000 Btu/hr heater at the radiator inlet increased TC1 to the order of -50 °F at $f=0.75$ (preset bypass fraction). But, the Coolanol inlet to HX1 (TC2) drops about 10 °F and overchills water heading for the cabin exchanger. Another possibility is to change the water flows WW1 and WW2 at the outlet of HX3 (bus interface). In the earlier calculations, WW1 was fixed at 114 lb/hr. This value was increased while WW2 decreased and improvement was noted. TC1 climbed to -58 °F with WW1 at 320 lb/hr. However, this appears to be as high as TC1 will rise with this scheme.

Very good results were realized by employing a regenerator heat exchanger between the radiator inlet and outlet streams, figure . This concept, including the deletion of radiator bypass, was documented in technology proposed for Skylab (6). The effect of a regenerator is to permit total flow through the radiator at all times, thus minimizing the heat load required to prevent freezing.

TABLE III

MODEL SOLUTION: Q2 = 1000 Btu/hr, 50% LOAD

St	Input	Name	Output	Unit	Comment
	1000	q2		Btu/hr	cabin air load HX2
		tw2	49.370066	F	water out HX2
		twl	40.598137	F	GUESS water out HX2
	88	fa		cfm	cabin air flow
	16	pa		psia	air inlet to HX2
		tal	70.617136	F	GUESS air in to HX2
		ta2	60.937710	F	air out HX2
		dt2	20.790021	F	log mean del t, HX2
		qrn	3372.9746	Btu/hr	net rad heat (+=out)
	366	wc		lb/hr	Coolanol flow
	.45	cpc		Btu/lb-F	Coolanol heat cap
		tc2	60.479505	F	GUESS Cool. inlet
	40	tc3		F	control valve set pt
		qrr	4081.9746	Btu/hr	heat reject by rad
	709	qra		Btu/hr	100%=1418 LOAD
		tr	25.326806	F	avg rad temp
	24	ar		sq ft	radiator area
		tcl	-9.825893	F	Cool. outlet rad
		f	.70870650	none	fract flow bypass rad
		ql	3372.9746	Btu/hr	HX1
		dtl	3.2683862	F	log mean del t, HX1
		tw4	70.185633	F	water out HX3
		q3	2749.5255	Btu/hr	HX3
	520	ww5		lb/hr	water flow
	1	cpw		Btu/lb-F	water heat capacity
		tw6	75.473182	F	GUESS water in HX3
		tf2	69.999137	F	facility out temp
	550	wf		lb/hr	facility water flow
	65	tfl		F	facility inlet temp
		tw3	58.356909	F	water out CP 1
	1024.5	qcpl		Btu/hr	100%=2049 LOAD
	114	wwl		lb/hr	water flow HX1
		tw5	67.592412	F	water to CP 2-5
	406	ww2		lb/hr	water flow CP 2-5
		qcp	4098	Btu/hr	CP 2-5 total load
	1024.5	qcp2		Btu/hr	100%=2049 LOAD
	1024.5	qcp3		Btu/hr	"
	1024.5	qcp4		Btu/hr	"
	1024.5	qcp5		Btu/hr	"

This claim was tested by revising the computer model to evaluate the regenerator performance theoretically without radiator bypass. Example calculations (figure 4) showing the effect of the regenerator vs. radiator bypass and other hardware configurations are the subject of an MSFC-NASA memo (7). (It is to be noted, however, that these calculations were based upon the old loop configuration as presented in the 1985 report (4)). This memo also considers the use of a combined regenerator and regenerator bypass valve. The regenerator would be operative at low heat loads, but it would be bypassed at high loads in order to gain the efficiency advantage of a hotter radiator.

HEAT PIPE SIMULATION

A. THEORY

Originally, heat pipes were simple tubes with an annular wick next to the inside tube wall. Limitations in heat flux, capillary pressure, pressure drop in liquid, and pressure drop in vapor motivated development of alternative configurations. A generation improvement was discovered by separating the vapor and liquid channels, finding finer wick structures, and machining axial or circumferential grooves to contain liquid in the evaporator and to enhance capillary pumping. A step towards high performance was suggested by Jim Owen of Structures and Propulsion Laboratory, MSFC. His design consisted of 6 vapor channels surrounding a central liquid channel, with a fine slot connecting each vapor channel to the liquid (figure 5). Heat would enter the outer tube wall, conduct into circumferential (or axial) grooves, and cause evaporation. The hypothesis was that special techniques, e.g., electrochemical or laser etching, could be employed to create very fine slots connecting vapor and liquid channels. If very fine slots and fine grooves were achievable, a higher capacity heat pipe would result, possibly even applicable to ground applications.

Liquid in a vapor-liquid-interconnecting slot has a meniscus which creates a higher pressure in the liquid as it evaporates (assuming perfect wetting):

$$P_V - P_{LS} = 2\sigma / w_S \tag{1}$$

where

P_V = vapor pressure, N/M^2, evaluated at T

P_{LS} = liquid pressure in slot, N/M^2

σ = surface tension, N/M

w_S = slot width, M

T = evaporator temperature, K

Liquid will also be situated in the grooves on the wall of the vapor channel. Assuming a v-shaped circumferential groove, the liquid-to-vapor pressure difference is:

$$P_V - P_{LG} = 2\sigma / w_G \tag{2}$$

where

P_{LG} = liquid pressure in groove, N/M^2

w_G = maximum groove width, M

Vapor leaving the evaporator undergoes frictional pressure drop (ignoring inertial and compressible effects), assuming laminar flow:

$$P_V - P_{VC} = 32\mu_V m_V L/(\rho_V A_V D_{HV}^2) \quad (3)$$

where P_{VC} = vapor pressure in condenser, N/M²

μ_V = vapor viscosity, kg/M - s or N-s/M²

m_V = vapor flow, kg/s

L = effective heat pipe length, M

ρ_V = vapor density, kg/M³

A_V = vapor channel cross section, M²

D_{HV} = hydraulic vapor diameter, M

The effective heat pipe length, L, is taken as:

$$L = \frac{1}{2} L_E + L_A + \frac{1}{2} L_C \quad (4)$$

where L_E, L_A, L_C are evaporator, adiabatic, and condenser lengths, respectively, M

The hydraulic diameter is related to the hydraulic area by:

$$D_{HV} = 4 A_V / WP_V \quad (5)$$

where WP_V = vapor channel wetted perimeter, M

For the liquid channel, equation (5) simply is changed to L subscript instead of V.

Condensation occurs at constant pressure with no surface curvature, so:

$$P_{VC} = P_{LC} \quad (6)$$

where P_{LC} = liquid pressure in condenser, N/M²

Liquid, returning to the evaporator via its channel has a frictional drop:

$$P_{LC} - P_L = 32\mu_L m_L L/(\rho_L A_L D_{HL}^2) \quad (7)$$

where the L subscripted variables are defined analogously to the V subscripted variables, and

P_L = liquid pressure in channel adjacent to evaporator

Liquid rises in the slot to enter the evaporator. This frictional pressure change is:

$$P_L - P_{LS} = \mu_L (L_S + W_S)^2 H_S (f \cdot R_E) m_L / (2 L_S^2 W_S^2) \quad (8)$$

where L_S = slot length (usually equal to Le), M

H_S = slot height (thickness), M

$f \cdot R_E$ = aspect ratio parameter, = 24 for $W_S / L_S \simeq 0$

If the number of vapor or liquid channels is not unity, then:

$$m_L = m_V \cdot NVC \quad (9)*$$

where NVC = # vapor channels

Finally, the heat flux is given by:

$$Q = m_L \cdot \lambda$$

where Q = axial heat propagated through the device, Watt

λ = heat of vaporization, J/kg

Equations 1 through 9 can be solved algebraically for the unknown pressures and heat flux given the evaporator liquid temperature, T, the heat pipe parameters, and liquid and vapor properties. The condenser temperature is fixed by the vapor pressure in the condenser, Pvc.

B. PROGRAM DESCRIPTION

A computer program was written to solve for unknowns as described above. Originally, the computation was done in BASIC to prove the concept. Only 30 lines of code were required to solve for pressures and Q, given all fluid properties and heat pipe geometry as inputs. At this writing, the program is being converted to FORTRAN for use on an IBM-AT. Regression equations (8) for fluid properties are included in subroutine PROP. Water, ammonia, acetone, and Freons will be included. Program documentation will be available through EP44.

*The above analysis assumes a single liquid channel.

C. CASE STUDIES

The 6-vapor channel design with a central liquid artery was solved for five cases with water as the fluid. Typical heat pipe parameters were chosen from the recent literature (9):

Total length = 2.03 M	Le = 0.305 M
Effective length = 1.575M	La = 1.115 M
Ws = 0.00079 M	WG = 0.00025M
Hs = 0.003 M	Lc = 0.610 M

Channel dimensions were taken from Jim Owen's design:

Dhv = 0.0127 M	Dhl = 0.01016 M

Outer tube diameter (surrounding vapor channels) = 0.0381 M

The five cases were:

	T, K (^{o}C)	Ws, M	Wg, M	Comment
1.	313 (40)	0.00079	0.00025	Base case
2.	313 (40)	0.000395	0.00025	Ws is halved
3.	373 (100)	0.000395	0.00025	T is increased
4.	373 (100)	0.00000395	0.0000025	5 psi capillary pumping
5.	373 (100)	0.00000395	0.0000025	Hs = 0.00003 M

Given the temperature of liquid in the evaporator grooves (T), the saturated liquid pressure is found from regression equations (or steam tables) and then all other variables are solved by the program. The results are given in table IV. At 40 ^{o}C, a water heat pipe is below optimum temperature. Peak axial heat flux at that temperature is 68.6 kw by theory. Since in case 1 the predicted heat flux is only 3.2 kw, this means that the groove and slot dimensions are sub-optimal also. Halving the slot width, case 2, increases axial heat flux to 4.8 kw, while lowering the priming pressure (Pls - Plg, cm water) from 3.9 to 2.1 cm water. (These pressures and the model are calculated for zero gravity). Case 3 has the smaller slot, but the temperature is raised to 100^{o}C. This raises the axial heat flux to 16.1 kw and further lowers priming pressure to 1.8 cm water. Proceeding to the ultimate goal, case 4 has a slot and groove combination (reduced by 100x) which produces about 5 psi capillary pressure (47,200 N/M^2), but pressure drop in the slot is so large that the liquid flow is retarded and Q drops to 0.76 kw. To reduce the slot pressure drop, the

TABLE IV

CASE STUDIES OF HEAT PIPE SIMULATION

(Pressures are N/M^2)

CASE NUMBER

	1	2	3	4	5
T, K	313	313	373	373	373
Tc, K	314	314	372.7	384	384
Plg	7500	7500	100,000	100,000	100,000
Pv	8060	8060	100,472	147,200	147,200
Pvc	7887	7713	100,193	147,199	143,836
Plc	7887	7713	100,196	147,198	143,835
Pl	7883	7706	100,174	147,197	143,588
Pls	7883	7706	100,173	117.327	117,327
ML, kg/s	0,0013	0.0020	0.00700	0.00716	0.02942
ΔPvap	172	347	279	2	3364
ΔPliq	4.6	6.9	22	1.0	247
Q,kW	3.24	4.84	16.1	0.76	67.2
QL, kW M	5.1	7.6	25.3	1.2	106
Prime Δ P	3.9	2.1	1.8	177	177
Turbulent	Vap.	Vap.	Vap.,Liq.	None	Vap.,Liq.

height (Hs) was reduced by a factor of 100 (to 0.00003 M) in case 5, resulting in Q rising to 67.2 kw. Hence, to reach a capillary pumping level on the order of 5 psi, very special groove and slot dimensions are required, e.g., Ws = 3.95 micron, Wg = 2.5 micron, and Hs = 30 micron. At this point, manufacturing techniques may be limiting, although it may be possible to use 2000-mesh screen to produce the desired 5 psi capillary pressure.

CONCLUSIONS

1. The thermal loop is constructed and preliminary checks are done. (Data measured at Lockheed is not available at this time.) The tests appear to have been successful.

2. Simulations of the loop have shown that it is well designed for its purpose. It may have a low-load freezing problem, and the radiator may be too small for high load situations.

3. Heat pipe calculations for 6-vapor channels with an external artery clearly show the effects of slot and groove width and temperature on heat pipe performance. The computer program is available on floppy disc for IBM-compatible PC's.

4. A heat pipe capable of 100 kw flux at 5 psi pumping level will be feasible only if wick or screen material is developed with dimensions of pores and thickness in the micron range.

NOTE: Subsequent work by the author has uncovered the existance of turbulent flow in the vapor channels (cases 1, 2, 3, 5) and in the liquid artery (cases 2, 5). The results given in table IV reflect turbulent equations where appropriate for flow and pressure drop. However, equations 3 and 7 above are written only for laminar flow. Turbulent equations may be found in fluid mechanics texts or reference (8). The computer program has been ammended to check for and handle both laminar and turbulent flow.

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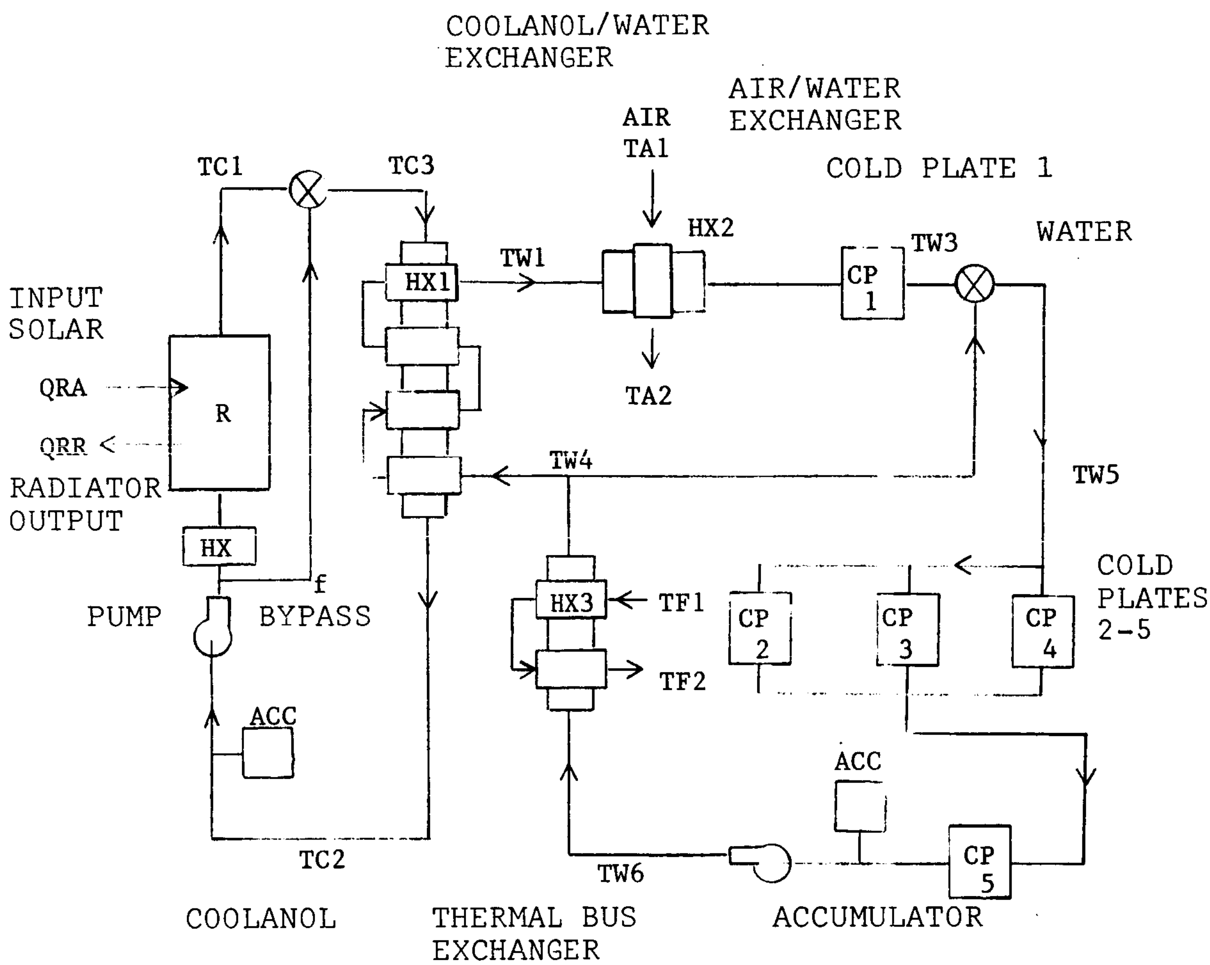

Figure 1. Schematic of thermal loop with variable symbols corresponding to TK!SOLVER model.

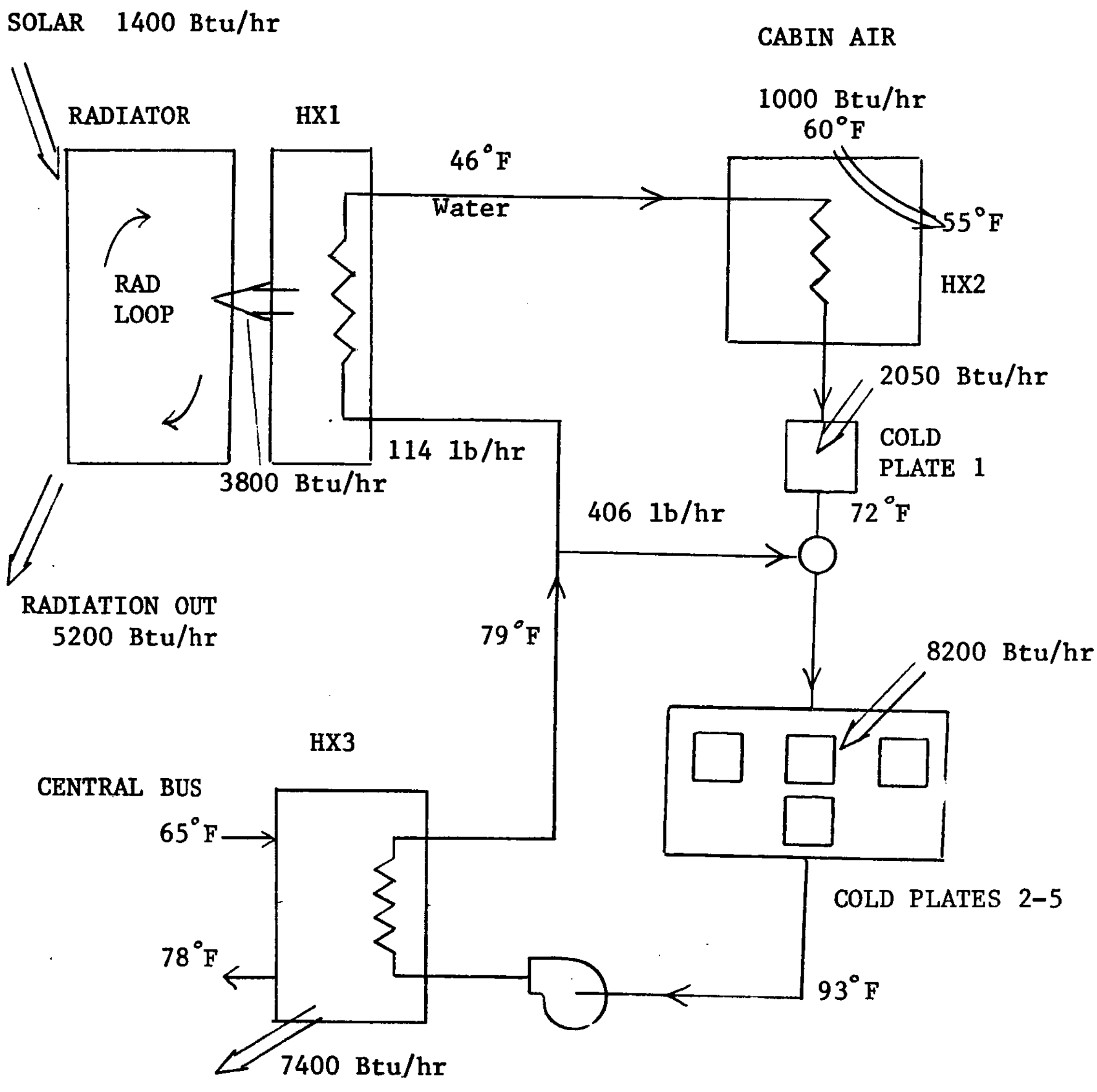

Figure 2. Typical heat flows in water and radiator loops (100% load).

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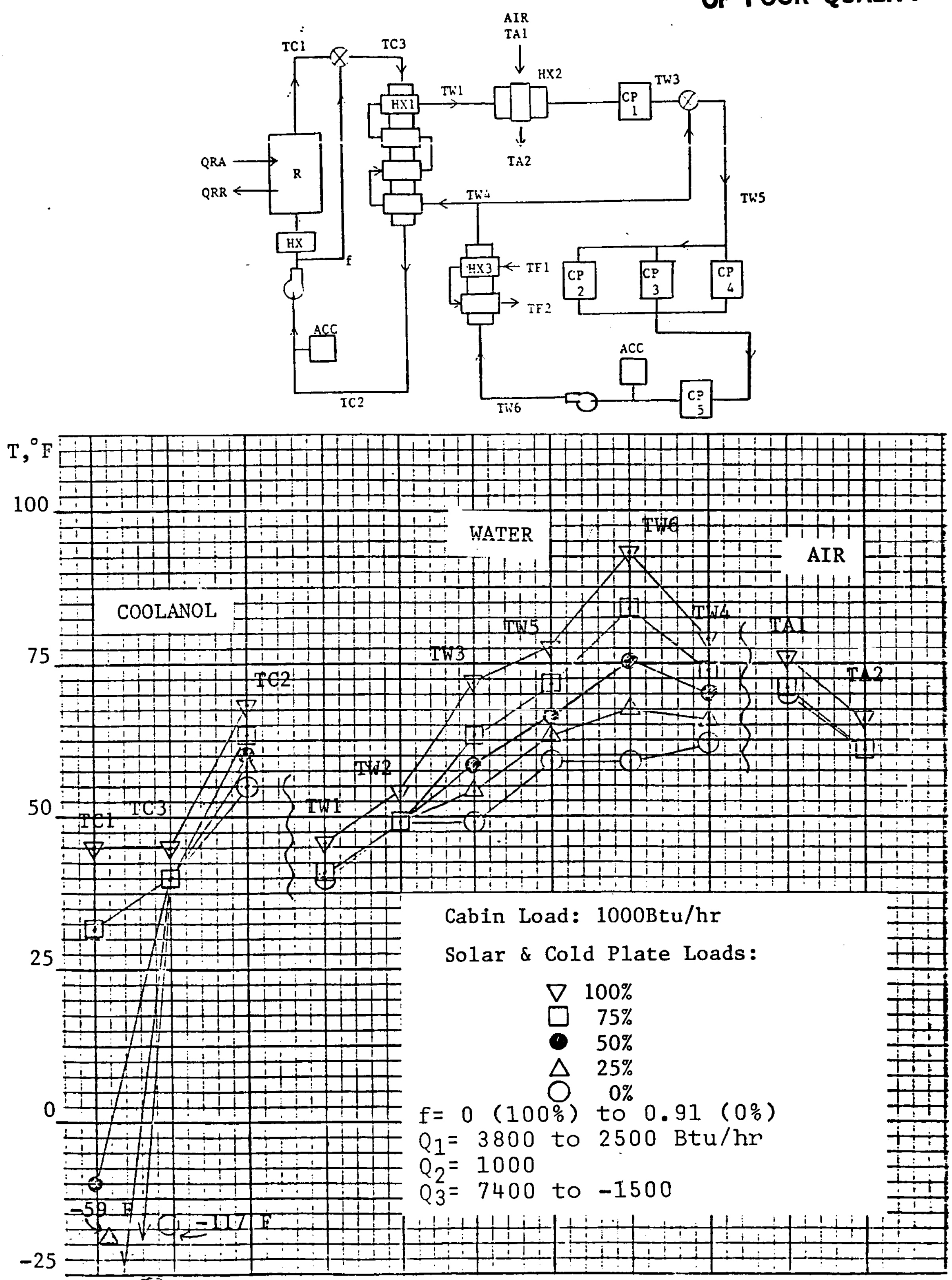

Figure 3. Thermal test loop constructed by Lockheed, Huntsville. Simulation of heat load effect on loop temperatures.

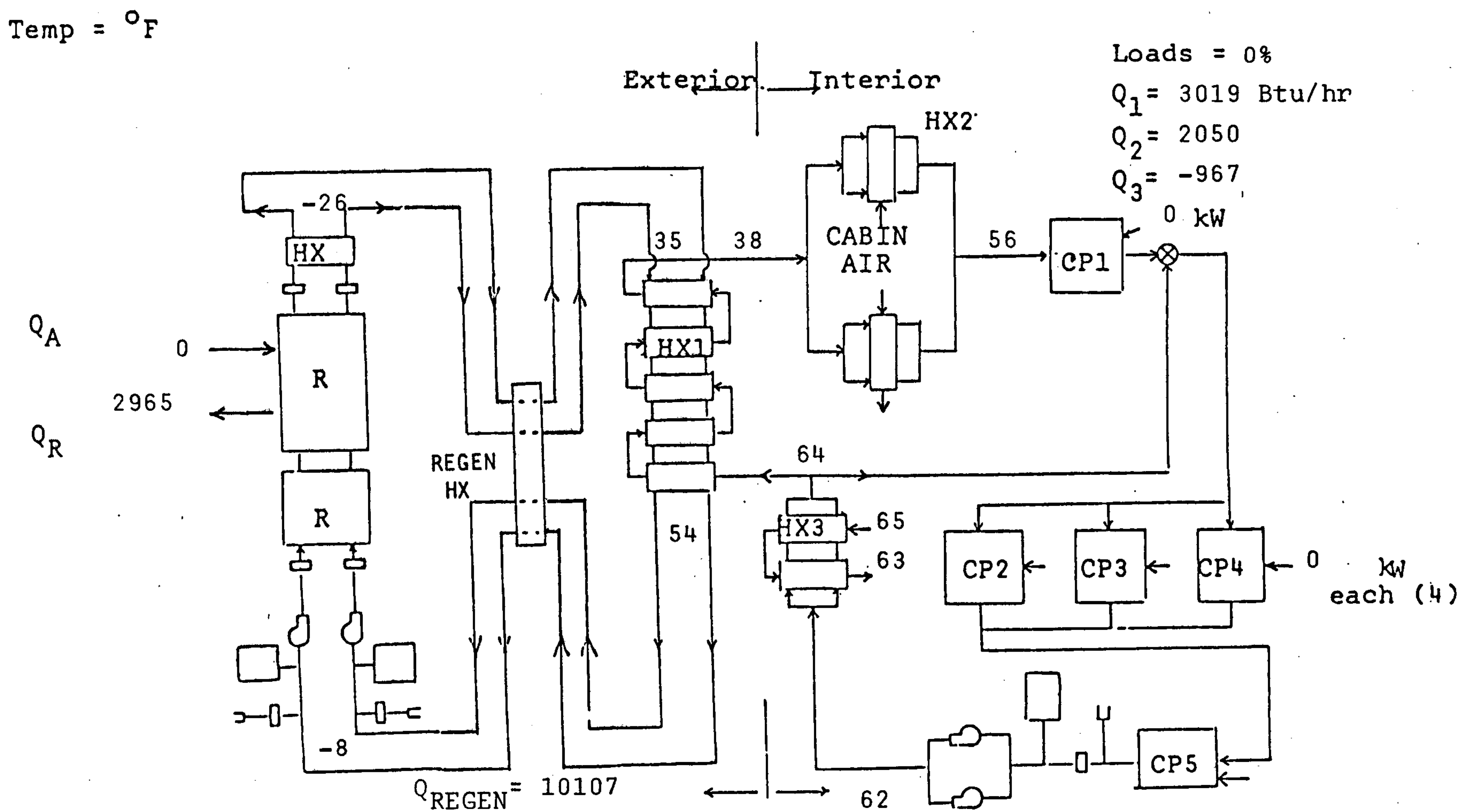

Figure 4. Regenerator heat exchanger model results for 0% load based on 1985 hardware.

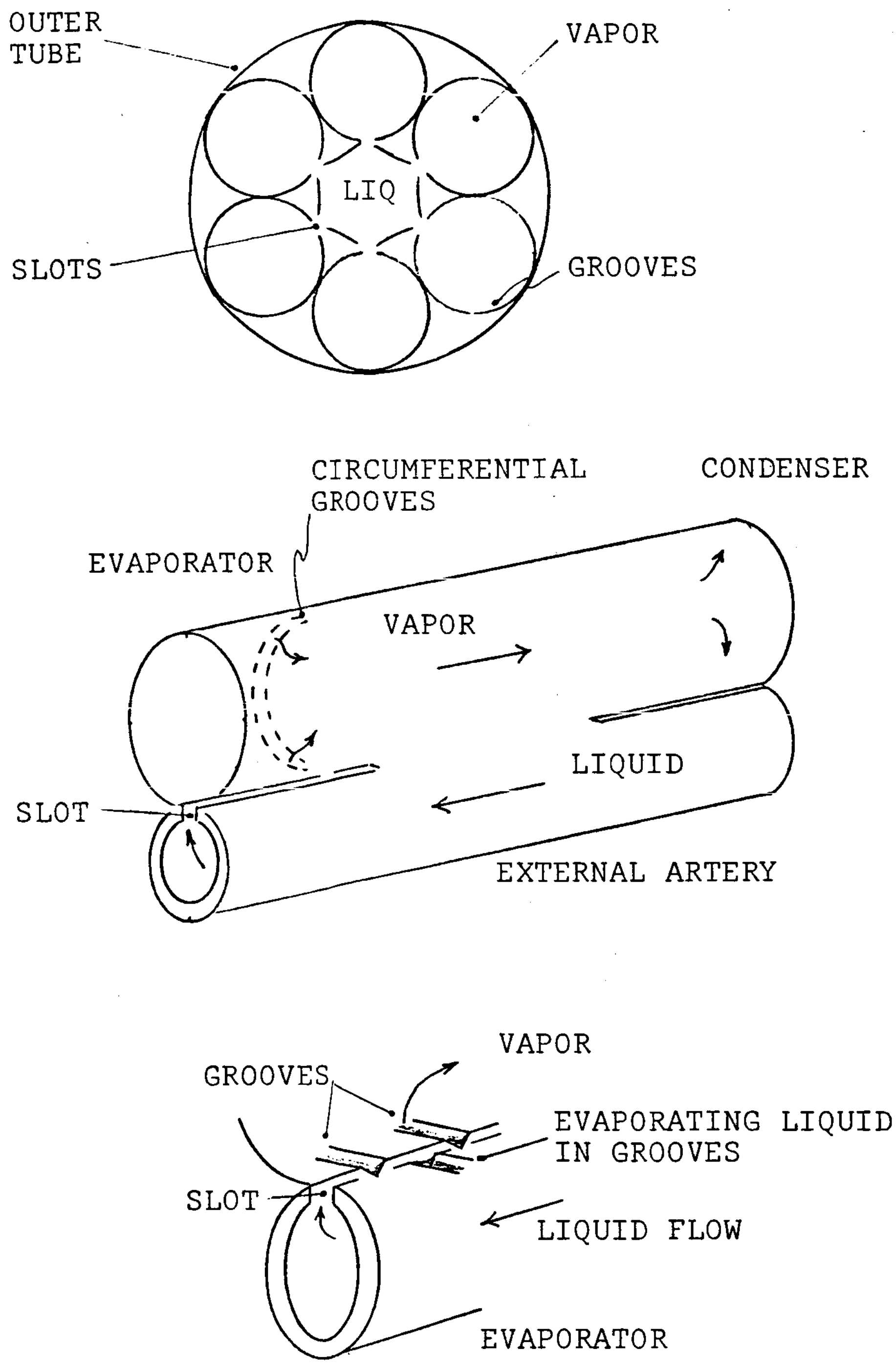

Figure 5. Top: 6-vapor channel heat pipe with external liquid artery. Middle: external artery with circumferential grooves. Bottom: blowup of slot and grooves.